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CORRELATION OF FATIGUE DATA TO DETERMINE STRESS CONCENTRATION FACTORS IN 76S-T ALUMINUM ALLOY

Thesis by

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PASADENA, CALIFORNIA

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SUMMARY

The basic problem of this particular investigation was to determine the stress concentration factors present at a shoulder in the root section of a model propeller blade manufactured from 76S-T aluminum alloy. The particular geometric section considered is found in the type propeller blades such as will be installed soon in the Southern California Cooperative Wind Tunnel, Pasadena, California. These blades were manufactured from forgings of 76S-T by the Hamilton Standard Propeller Division of the United Aircraft Corporation. Stress concentration factors were determined for what will be called the "critical section" of the blade. This section is understood to mean the cross section of minimum area which is located immediately above the stress-raising fillet at the junction of the blade itself and the root flange.

A secondary purpose of the investigation was to correlate these data with two other experiments that had already been carried out for this particular blade shape. These earlier experiments determined the magnitudes of surface stresses in the critical section, one utilizing a full-scale, three-dimensional model, and the other a two-dimensional full scale model one-inch thick. Both of these experiments utilized static tension loads, the loads being applied over the upper surface of the flange, exactly as they are assumed to act when the blade is rotating in the wind tunnel. Also, an attempt was made to compare the data obtained in this investigation with other fatigue tests of a more general nature.

The tests made in this present investigation were all of a fatigue nature, in which one-tenth scale models of the actual blade were used as fatigue specimens. Two types of loading were used. The first consisted of a cyclic loading between varying upper tension limits to zero stress, and the second consisted of a constant upper tension stress limit with varying minimum stresses.

Tests were conducted using an upper tension nominal stress limit of approximately 28,000 psi and were extended to tension stresses of lower values which gave a fatigue life of more than 15,000,000 cycles. Due to the slow rate of loading (2,500,000 per day) in the Sonntage Universal Fatigue Testing Machine, the investigation was not continued to the generally accepted value of 500,000,000 cycles which marks the upper cyclic limit of fatigue investigations.

It was found that stress concentration factors determined from this type of fatigue testing came surprisingly close to those determined from the full-scale, three-dimensional, static tests.

The selection of the type loadings used was not entirely arbitrary. The stress cycle which varied from zero to a peak tension value and then to zero again would closely simulate the loadings in a start-stop cycle; while the superposition of a cyclic stress upon a steady tension load would result in approximately the type of loading experienced by the blade when it was rotating at a constant angular speed under aeroelastic axial forces.

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I. INTRODUCTION

The mechanism of fatigue failure in various components of machinery has long been one of the great unsolved mysteries of the physical world in which we live. Since no analytic solution exists, the fatigue life of a particular material has, at the present time, to be determined experimentally. Even experimental results, coming from carefully controlled tests, are sometimes misleading and incongruous and, therefore, must be carefully analyzed before being used in the actual design of machinery. Metallurgical science, confronted with demands for alloys of increasing tensile strength, hardness, etc., is similarly challenged with the necessity of keeping the increase in the resistance to fatigue in proportion to the improvements in other characteristics. It has, therefore, been necessary to conduct countless experiments using specimens of varying sizes, shapes, and material, under different conditions of loading, in order to accumulate a vast amount of data, all of which someday may lead to a vital clue to the factors influencing fatigue properties. It is hoped that this experiment may be one of the building blocks which eventually may pave the road to such understanding.

This project was conducted by the author, and Lieutenant John C. Kane, U. S. Navy, at the Structures Laboratory of the Guggenheim Aeronautical Laboratory at the California Institute of Technology, Pasadena, California, under the guidance of Dr. E. E. Sechler, during the period from 15 November 1949 to 1 May 1950.

This experiment, although of limited general value, had as its

purpose the investigation of the fatigue life, and hence the stress concentration factors present at a shoulder in the root section of the type propeller blade such as is to be installed soon in the Southern California Cooperative Wind Tunnel. The original propeller blade installed in the wind tunnel was manufactured from 25S-T aluminum alloy and had as a distinguishing design feature a 3/16-inch radius circular fillet at the root flange. After 3000 hours of operation, one of the blades in the propeller assembly failed at the root. A picture of the failure is shown in Figure 1. The failure consisted of a complete fracture at the fillet which closely resembled a typical fatigue type failure. Before deciding on the design of a replacement blade, several different shapes were considered. Static two-dimensional tests were run on proposed designs, all of which attempted to determine which of them would minimize the concentrated stresses at the fillet and yet meet the other geometric specifications dictated by tunnel arrangement. The new designs utilized flanges of varying depths, and fillets of various circular and elliptic shapes. The final design selected was one having a 3½-inch deep flange, and a 0.75-inch circular fillet. It was determined to conduct the present investigation to correlate the stress concentration factors as determined from static tests with those determined in an investigation utilizing cyclic, or fatigue, loading.

Because of the unconventional method of applying the cyclic loading to the specimen and because of the many parameters influencing the tests such as specimen fixity in the holding jig, presence of bending stresses, and slight dimensional inconsistencies among the

several specimens tested, etc., the results as published herein are not meant for exact quantitative use in the design of future propeller blades, but are offered merely as an indication as to how further fatigue tests of this nature may be conducted and in order to give the designer a general correlation between statically and cyclically determined stress concentration values.

II. EQUIPMENT AND PROCEDURE

Specimens

All of the specimens used were manufactured from an original unfinished 76S-T propeller forging. The specimens were cut from the material taken from the widest and thickest part of the blade and were machined with the axis of symmetry running "with the grain", i.e., parallel to the long dimension of the forging.

The specimens were constructed to one-tenth scale and are shown in detail in Figures 2 and 3. The dimensions were obtained from the original SoCalCWT blueprint No. SK-16000, Change A. Because of the small dimensions involved on the model, certain minor features of the original blade could not be accurately incorporated therein. These include such things as the taper of the central hole, various attaching countersinks and holes, etc. It is felt, however, that none of these features as omitted in the specimens had any significant effect on their fatigue life.

The central hole drilled in the specimens was of constant diameter. This diameter was arrived at by determining the size of the hole in the original blade at its minimum cross sectional area.

Because of the great importance of surface finish in the fatigue life of a specimen, (cf. Ref. 1), the surface of all models was polished to approximately 5 microinches. Great care was exercised during the tests to insure that no tool marks, scratches, or other surface imperfections were introduced which might cause localized

surface stress concentrations and a resulting premature fatigue failure. The central hole was first drilled with a standard drill, then was hand polished with levigated alumina on a small wooden dowel. Examination of failed specimens did not indicate any fatigue cracks originating from imperfections in the surface of the central holes. Therefore, it is assumed that the above process was entirely satisfactory for the requirements of this project.

As a check on the dimensional accuracy of the machining operations a specimen was chosen at random and checked in a Jones and Lamson Optical Comparator. The results of this check were good, and they are shown in Figure 4.

Method of Specimen Fixture During Testing

The specimens were mounted in a holding jig designed by the investigators and constructed in the GALCIT machine shop. It is shown in detail in Figures 5 and 6. The jig approximated the fixture of the propeller blades in the wind tunnel except that no provision was made for the ring clamp that holds the blade a few inches above the fixture on the flange. Since the criterion of this investigation was for pure axial tension with the absence of bending stresses, it is felt that the omission of this clamp was of no great importance. The holding jig applied the load to the upper surface of the flange exactly as it is applied by the restraining forces on the operating blade. The loaded surface of the model is an annular area on top of the flange, the mean radius of which is 0.355 inches with a thickness

of 0.375 inches. Due to the small dimensions involved it was felt that it might not be practicable to load the specimen over such a small area. As a result, one test specimen having a root diameter of 0.318 inches and an annular loading area, the thickness of which was approximately double that of the exact scale model, was constructed and tested. Comparing the fractures and appearance, after failure, of the enlarged specimen to that of the exact scale model failed to indicate that the smaller exact scale specimen was not entirely satisfactory; therefore, the exact one-tenth scale models only were used during the remainder of the investigation.

The loads, all of which were tensile, were actually introduced into the model by the contact of the cap ring of the holding jig as shown in Figures 5 and 6. The first cap ring used was manufactured from mild steel and was 0.250 inches thick. After approximately 30,000,000 test cycles, one of the original chromium-plated steel machine screws used to fasten the cap ring failed from fatigue. Microscopic examination of the fracture showed a serious occlusion in the material at the point of failure. After this particular screw had failed, the loading cycle continued a few seconds until the machine could be stopped. During this time the high loads imposed on the unsupported portion of the cap ring caused it to warp badly, making it unfit for further use. In order to increase the rigidity of the entire jig a replacement cap ring was made from high strength steel with a thickness increased to 0.3750 inches from the original 0.250 inches. The holding screws were replaced with $\frac{1}{4}$-inch Allen bolts. After these changes were made the tests proceeded without further trouble in the

holding arrangement.

Method of Testing and Stress Measurements

The general set-up for the experiment is shown in Figures 7 and 8. The strain gage wiring is shown in Figure 10.

Loads introduced during fatigue testing were measured by three type A-1 Baldwin Southwark electrical strain gages mounted axially on the surface of the specimen. The location of the gages is shown in Figure 2. The gages were mounted by the following practice: the surface was roughened with No. 00 sandpaper, cleaned with solvent and then air-dried. The radial location of gage centerlines was accurately determined by placing the specimens in a lathe and then by dividing the circumference into 120 degree arcs. Gages were fastened with radio service cement, then were wrapped with several turns of carpet thread to hold them in place during drying. Uniform drying was accelerated by baking the specimens for eight hours under infra-red lamps at a mean temperature of 150° F. This process completed the polymerization of the adhesive. Had this not been done, the cement would have remained in an unstable condition for several days and any readings taken during that time would have been inaccurate.

Before commencing the actual fatigue tests, an attempt was made to verify the gage factors as published by the manufacturer. Extensive tests on the performance of the A-1 type gage were conducted by Campbell as described in Reference 2. The results of Campbell's tests on eight, A-1 gages, selected from random lots, are shown in the following table:

Gage No.	Resistance in ohms	Gage Factors incr. strains	decr. strains
1	120.7	2.028	2.065
2	120.5	2.044	2.060
3	120.5	2.056	2.064
4	120.5	2.064	2.070
5	120.5	2.037	2.074
6	120.4	2.054	2.060
7	120.5	2.058	2.076
8	120.5	2.060	2.070
Mean	120.51	2.0501	2.066

These tests closely corroborate the factor of 2.04 as published by Baldwin Southwark for the lot (Lot. no. "MZ") of strain gages used in this experiment. The gage factors of three arbitrarily selected A-1 gages as determined experimentally by the investigators on this project averaged 2.063, and were arrived at by the following procedure: An arbitrarily selected scale specimen was mounted in its holding jig and loaded in static tension in a Riehle Beam-balance Testing Machine. The strain gages were wired as shown in Figure 10. Loads were applied from zero to 2000 lbs. in increments of 500 lbs. Gage voltages were measured by a Leeds and Northrup Potentiometer. Prior to testing, all electrical connections between the gage wires and the lead wires were checked with a vacuum tube ohmmeter for shorts and high resistances. All gages except one, which was replaced, were found to be satisfactory in this respect.

Approximately ten complete cycles of loading were necessary before the gage readings settled down to uniform increments as the loads were applied and removed. Due to the design of the Riehle machine it was impossible to remove all of the bending in the specimen

during these tests. This was corrected by running three complete tests. Between each run, the specimen and jig were rotated 120° in the end clamps of the machine. Gage voltages indicated that the plane of bending was constant with respect to the base of the machine. The effect of this bending was cancelled by taking the average of the voltage increments in the three different positions.

By using the formula developed in the Appendix, it was possible to calculate the gage factors for these three gages as long as the applied stresses, the potentiometer deflections and the battery voltage were known. Results of these tests indicated that the factors for the three gages selected were 2.07, 2.07, and 2.05. Combining these results with those determined by Campbell and with those published by the manufacturer, the strain gage factor for all gages used during the remainder of the investigation was taken as 2.05.

The power supply for the electrical strain measuring circuits was a standard three-cell, 6 volt, wet storage battery.

Stresses during cyclic loadings in the fatigue tests were measured as follows: a testing panel, designed and built by Mr. Marvin Jessey of the Guggenheim Aeronautical Laboratory, permitted the introduction of a fictitious stress trace into a Heiland Recording Oscillograph which gave the same indication on the recording paper as an applied end load of 2000 lbs. acting on the specimen would give through the three strain gages. The introduction of this fictitious, or calibration, trace was accomplished in the following manner: By disconnecting the three strain gages from the testing panel and

substituting a precision variable resistance in their place, it was possible to determine the resistance which, when placed in the circuit instead of a gage, would cause the same potentiometer deflection (in millivolts) as if a 2000 lb. load were applied to the specimen. Precision wire-wound resistors of the above values were then wired into an arbitrary circuit of the testing panel. When a pushbutton switch was closed, all other circuits were deënergized except the one containing the fixed resistors. This circuit, when fed directly into the Heiland, caused the fictitious 2000 lb. trace on the recording paper.

An additional useful feature was built into the test panel by incorporating a variable resistance in series with each active gage. By adjusting each of the three resistors when the specimen was unloaded, the potentiometer dial could be completely zeroed, thereby making it unnecessary to subtract original zero reference readings when actual strains were measured during the tests.

The fatigue tests were conducted in a Sonntag Universal Fatigue Testing Machine, Model SF-1-U, Serial No. 492004. The standard multiplying fixture was used in which the values of dynamic and static loads as set on the machine indicating dials were multiplied by 5. Cyclic loads were set by adjusting the eccentric length of the rotating weight, whereas static preloads were varied by changing the tension on the main support springs. A complete description of these adjustments is given in Reference 5. Spring tension was measured with a built-in micrometer which indicated spring deflection. The spring constant for this particular machine was that .001 inches represented

4.17 lbs. of static pre-load. According to the machine specifications, all machine load-measuring devices were accurate to plus or minus 2 per cent. Subsequent strain gage checks verified this contention.

Specimens were mounted in the machine by means of the holding jig previously described. After the jig was mounted loosely in the machine (Figure 9), all strain gages were zeroed. The grip holding screws (Piece No. 9, of dwg. No. 50497-L of Reference 5) were then tightened, leaving the upper column support ring loose. Because the tests were run in tension only, the spherical compression seats (piece No. 5 of the same drawing) were removed. The desired preload was then set by adjusting the spring tension and the cyclic load set by varying the eccentric arm. After the machine was run a few cycles to permit the ball seats to become free, it was stopped and zero readings were checked again. It was assumed that uniform voltage increments for the three gages indicated no bending, at least when only the static load was acting. Major variations in these check readings were eliminated by adjusting the tension on the twelve grip screws. Final gage uniformity was achieved by adjusting the tension on the studs that tighten the upper column clamp ring. Before the tests were actually started, all three gage readings were brought within plus or minus 20 microvolts of each other.

When all these preliminaries had been completed, the machine was started and the test begun. Dynamic measurements were then taken as follows: The switches for all three gages on the testing panel were set to the "GAGE" position. In this position only the zero readings were transmitted to the Heiland and therefore only a straight,

undeflected trace appeared on the paper. This trace represented the zero load level. The switches were then set to "HEILAND" which introduced the sinusoidal traces representing the cyclic part of the load. These cyclic stresses were displaced from the zero load level by an amount corresponding to the static preload. After these readings were made the machine was shut off and another "HEILAND" trace was made. This trace was another straight line, displaced from the zero load level by an amount corresponding to the static preload. Next, the pushbutton switch on the panel was depressed. This introduced the previously described calibration trace corresponding to the arbitrary load of 2000 lbs. The recording tape for the run was then complete. Using this last trace as a measuring scale, all other loads appearing on the record could be measured quantitatively. A typical Heiland record for one run is shown in Figure 13. Signals of equal amplitude from all gages indicated the optimum condition of no bending. In every test run the amplitude variation between gages was less than 3 per cent.

Methods of Loading

The experiment was divided into four parts. Part I consisted of the preparation of a basic R. R. MOORE rotating beam S-N curve for 76S-T. Standard R. R. MOORE test specimens were fabricated according to the specifications in Reference 4. Fourteen of these specimens were used, all of which were finished to a high polish in the fracture area. They were run to failure in four separate R. R. MOORE Testing Machines, serial numbers 266, 268, 270, and 271, all of which ran at a

nominal speed of 10,000 RPM. The endurance limit, or stress at which the specimens did not fail after 500,000,000 cycles, was determined to be approximately 22,000 psi. This corroborates the values given by several other investigators. The final S-N curve is shown as curve No. 2 in Figure 14.

It is generally agreed that any fatigue data must be evaluated with respect to the type of loading applied to the specimens. The actual propeller models (in Part II) were loaded over their entire cross-sectional area, but in the R. R. MOORE type of loading only the outermost fibers in the section were subjected to the apparent stresses indicated on a R. R. MOORE S-N curve. As a result, it was decided to run a concurrent experiment in which un-notched specimens were fatigued by axial loads. A curve showing the results of this type of test for small, polished, unnotched specimens of 76S-T was supplied to the investigators by Mr. R. L. Templin of the Aluminum Company of America, but a few verifying points for the forging used in the construction of the propeller specimens were desired. Three smooth, un-notched specimens were tested in the Sonntag machine under cyclic loadings which went from zero to varying upper tension limits and then to zero again. The diameter of these specimens and those used by Alcoa were all less than one-half inch. These three check points fell well within the assumed scatter band of the Alcoa curve, (curve No. 1 of Figure 14). This experiment was conducted in the same machine and under exactly the same conditions as prevailed when the propeller models were tested.

The main part of the investigation (Part III) involved the actual testing of the propeller models. Nine scale specimens were

used of which two did not fail. This test utilized a cyclic loading of pure tension. The cycle commenced at zero, rose sinusoidally to a peak value, and then returned to zero. This loading was introduced by setting a static preload equal to one-half the desired peak value, and then by superimposing on it a completely reversed cyclic load whose half-amplitude was also equal to one-half the desired peak value. The resulting loading is similar to that shown in the Heiland record of Figure 13.

In order to determine the upper stress limit for this test, an assumed stress concentration factor of 2.0 in the critical section was used. Machine settings were adjusted to bring the apparent stress in the critical section to approximately 40,000 psi. As the tests progressed, the loads were reduced until the stresses were such as to give an endurance life of more than 15,000,000 cycles. The final S-N curve showing the results of this test is shown as curve No. 3 of Figure 14.

The last part, (Part IV), of the investigation consisted of testing four more specimens in a different loading schedule. A constant upper nominal stress limit of 28,000 psi in the critical section was selected. In the first specimen, the lower limit was zero; in the second, 5580 psi; in the third, 11,200 psi, and in the fourth, 16,840 psi. The results of this test (cycles to failure) are shown in Figure 15.

III. RESULTS AND DISCUSSION

The R. R. MOORE endurance limit of small, polished, un-notched specimens of 76S-T aluminum alloy was verified to be in the neighborhood of 22,000 psi.

The tests of three small, polished, un-notched specimens of 76S-T aluminum alloy in an axial-loading Sonntag Universal Fatigue Testing Machine (1800 cycles per minute) gave data points that corresponded almost exactly to those determined by Alcoa in similar tests of the same alloy.

The stress concentration factors existing in the critical section of seven model propeller blades loaded in tension were found to average 1.875. Stress concentration factor as used in this report is defined as follows:

$$SCF = \frac{\text{Apparent maximum stress}}{P/A}$$

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"Apparent maximum stress" is defined as the stress determined from a S-N curve for similar un-notched specimens as a function of "cycles to failure". "P/A" is defined as the applied axial load divided by the cross-sectional area at the critical section. The loading in these tests consisted of a constant static tension load upon which were imposed cyclic loads whose half-amplitudes were equal to the static load.

The cycles to failure determined in the tests of Part IV in which the upper tension stress limit was constant at 28,000 psi, and the lower tension stress limit was varied between zero and 16,800 psi,

are plotted in Figure 15. Because of the small number of specimens tested, no exact conclusions can be reached, but the results bear out the natural supposition that the number of cycles to failure decreases as the overall stress range increases.

Discussion of Results

Although it is impossible to separate and evaluate their individual effects, many sources of error are present in any fatigue test. The sources of error in this investigation which may tend to invalidate the test results come principally as a result of the unique method in which the actual loads were applied to the specimens. The great majority of fatigue investigations that have already been conducted by the engineering profession have consisted either of rotating beam tests or straight push-pull tests in which the specimens have been continuous pieces of metal held between two machine grips of one form or another.

Two possible sources of error inherent in all tests are (1)non-homogeneity of the material, and (2) small defects in the surface. It is generally accepted that fatigue failures are of a progressive nature, beginning where "the relatively highest stresses go to work on the relatively weakest parts". As a result, any localized stress concentrations such as would come from microscopic tool marks or interior occlusions would cause lower endurance limits than would be exhibited by theoretically homogeneous, perfectly smooth specimens. The effect of any non-homogeneity is felt to be entirely secondary in these tests, in that stress concentration factors were determined from

comparisons of tests of similar un-notched specimens. This is as it should be. Non-homogeneity is not only a possible factor, but it is inevitable. Small occlusions and surface irregularities are present in propeller blades manufactured under the most rigid specifications, so as a logical conclusion, any small scale studies not only should be, but must be made of the same material.

Another factor that cannot be overlooked is the possibility of small surface scratches in the neighborhood of the "critical section". The specimens were made by expert machinists working under the best shop conditions, but the small dimensions in the critical sections precluded microscopic surface surveys. It is, therefore, largely a matter of conjecture as to how serious any surface irregularities were.

Remembering the previously described criterion for establishing stress concentration factors, it is easily seen that assumption of uniform angular stress distribution over the critical section was probably very optomistic. This distribution would exist only in the absence of bending and in the presence of uniform circumferential clamping pressures. Many investigators have been disturbed by the difficulty in loading a fatigue specimen in pure axial tension or compression with the absence of bending, including the author of this report. Although the strain gages indicated that bending was reduced to a minimum at the beginning of the tests, there is no justification for assuming that such loading existed as the individual tests progressed. The jig holding bolts loosened repeatedly during the tests and had to be tightened periodically. This eliminated any possibility of constant flange pressures. Also, at the end of the investigation,

wear began to show on the clamping ring halves, especially on their ends. This factor alone would cause uneven circumferential pressures on the flange of the specimen. These factors, together with the inherent errors in electrical measuring equipment all must be considered in evaluating the results of these tests.

All of the foregoing present the unfavorable side of the picture and tend to indicate that all the data must be viewed with suspicion. However, the scatter band obtained, and the general succession of points plotted in Figure 14 indicate that the results are worthy of consideration when decisions concerning the geometry of the critical section of a propeller blade are being made.

Before the results of this investigation can be applied to the full scale propeller blade, the effect of relative sizes must be considered. No analytic relationship between endurance lives of small and large specimens of similar shapes is yet known, although many investigators have shown that a notch, or other stress-raiser, has a much greater effect on the actual large part than is indicated by laboratory tests of small scale models. Sachs, in Reference 6, found that the endurance limit at 10,000,000 cycles of a 3-inch diameter magnesium alloy propeller was 7000 to 10,400 psi, but when the diameter was increased to 4½ inches the endurance limit dropped to 4600 to 7000 psi.

The speed of loading appears to have little effect on the endurance of aluminum alloys. Tests conducted by the National Bureau of Standards (Reference 7) showed close agreement between fatigue tests conducted at 900 and 12,000 cycles per minute. Other investigators

have shown that this agreement is good up to and including 30,000 cycles per minute.

Correlation with Other Tests

The three-dimensional full-scale static tests of the same blade shape conducted by the Cooperative Wind Tunnel, as discussed in Reference 9, showed a stress concentration factor at the critical section of about 1.90. This value is in good agreement with the average factor of 1.875 as determined for in this investigation, and therefore indicates the usefulness of this type of fatigue testing. The loads in the static test were applied to the top of the root flange exactly as was done in the fatigue tests. The static stress concentration factors are defined as the ratio of measured stresses on the surface of the critical section to the nominal P/A stresses which are assumed to exist uniformly over that cross section.

As would be expected, the two-dimensional tests indicated much higher stress concentration factors. Reference 10 gives the ratio of measured surface stresses to nominal stresses in the critical section as being in the neighborhood of 3.18 for the two-dimensional model. Photoelastic investigations such as are reported on in Reference 8, of two dimensional axially loaded models having the same ratio of fillet radius/critical diameter as the propeller models, show a stress concentration of 2.13.

IV. CONCLUSIONS

1. The endurance limit of small, polished, unnotched specimens of 76S-T aluminum alloy is in the neighborhood of 22,000 psi when tested as rotating cantilever beams.

2. The high-stress portion of the S-N curve for 76S-T under axial push-pull loading, as published by the Aluminum Company of America, is essentially verified.

3. The stress concentration factor existing at the critical section of a one-tenth scale model of a wind tunnel propeller blade manufactured from 76S-T is in the neighborhood of 1.875.

4. Stress concentration factors established from fatigue testing are essentially lower than those determined from static tests of two-dimensional models.

5. Stress concentration factors determined from carefully conducted fatigue tests are of great value in designing the shapes of certain critical sections, in that fatigue testing makes use of alternating loads such as the actual full-scale part will experience when in service.

6. Stress concentration factors established from fatigue testing of one-tenth scale propeller models were essentially the same as those determined in static tests of full-scale blades.

7. There is an indication that stress concentration factors of small specimens are not dependent on geometry alone, but tend to decrease as applied loads over the cross sections are lowered. This is suggested by the relationship between the slopes of curves No. 1

and 3 of Figure 14.

8. In tests utilizing cyclic tension loads only, with a fixed upper stress limit, the fatigue life of a small, polished, propeller blade model of 75S-T decreases as the ratio of max/ min in the cycle increases.

9. Stress concentration factors determined from fatigue tests of small models should be used for the actual determination of design loads in the full-scale part only when the size effects for the material under investigation are known.

Recommendations for Further Study

1. Conduct similar tests in which the size of the specimen is the variable parameter.

2. Conduct similar fatigue tests of the model in the R. R. MOORE type of testing machine.

3. Investigate the effects of surface rolling, understressing, overstressing, and shot-peening on the fatigue life of similar specimens.

4. Continue any of the above investigations to approximately 100,000,000 cycles. Such investigations will not be practical unless an axial loading fatigue machine with a much greater speed than 1800 cycles per minute is developed.

5. Investigate the design of a fatigue testing machine in which the two specimen grips move within fixed slides, thereby eliminating bending and torsional interference.

6. If further tests similar to the one described in this

report are attempted, a redesign of the holding jig is necessary. Although a preliminary strength check of the jig was made, it was apparent that several of the components were over-stressed. The cap ring was not rigid enough, and there was no way in which uniform clamping pressures could be obtained. It would be desirable to make the cap ring fastening bolts come all the way through the base assembly. This would permit micrometer measurement of the bolts which in turn would simplify uniform bolt pressures. All parts of the jig should be made of high-strength steel with adequate heat-treatment.

7. Conduct a theoretical investigation of the stress distribution in the critical section by a relaxation method.

8. Make a three-dimensional photoelastic study of the model in which particular attention is paid to the portion of the base flange affected by the clamping pressure on the top of the flange.

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10. "An Experimental Study of the Stress Distribution in Two-Dimensional Fan Blade Specimens Under Static Load"; C.W.T. Report, December, 1949.

TABLE I

PROPELLER BLADE SPECIMENS
FATIGUE TEST RESULTS

$$\frac{\text{Min. stress}}{\text{Max. stress}} = 0$$

(Loads in lbs.)

Specimen No.	Machine Load			Heiland Trace Gage No.			Cycles to Failure
	Dyn.	Static	Total	1	2	3	
1	2500	2550	5050	5150	4960	5120	30,000
2	2500	2535	5035	No record			64,000
3	2430	2460	4890	4750	4780	4850	91,000
5	2250	2290	4540	4600	4460	4540	175,000
4	2000	2050	4050	4020	4110	4060	257,000
8	1980	2000	3980	no record			300,000
9	1960	1980	3940	no record			400,000
6	1880	1920	3800	3900	3850	3850	14,500,000 did not fail

TABLE I-A

(STRESSES)

All stresses in psi (tension)

Specimen No.	Machine Indicated (P/A)	Heiland Trace Gage No.			Variation from Machine Indicated (%)		
		1	2	3	1	2	3
1	27900	28450	27400	28285	3.0	1.8	1.3
2	27820	no record			no record		
3	27020	26240	26440	26800	2.8	2.2	1.0
5	25080	25410	24640	25080	1.0	1.7	neg.
4	22380	22210	22710	22430	1.0	1.5	neg.
8	21990	no record			no record		
9	21760	no record			no record		
6	20990	21550	21270	21270	2.6	1.3	1.3

TABLE II

RESULTS OF FATIGUE TESTS WITH VARYING MINIMUM STRESSES

Test Machine: Sonntag Universal Fatigue Testing Machine
Specimens: 1/10th scale propeller blade models
Surface Finish: 5 microinches
Speed of Loading: 1800 cycles per minute

Test Results

Specimen No.	Machine Load (lbs) Static	Machine Load (lbs) Dynamic	Max. Stress (psi)	Min. Stress (psi)	Stress Range (psi)	Cycles to Failure
1	2535	2535	28,000	0	28,000	30,000
2	3040	2030	28,000	5,580	22,420	60,000
3	3550	1520	28,000	11,200	16,800	816,000
4	4060	1010	28,000	16,840	11,160	did not fail at 35,000,000

TABLE III

R. R. MOORE FATIGUE TEST RESULTS

Material: 76S-T Aluminum Alloy
Machine: R. R. MOORE Fatigue Testing Machine
Machine Speed: 10,000 RPM
Specimen Design: Standard R. R. MOORE (Reference 3)
Surface Finish: 5 microinches

No.	Tare lbs.	Added lbs.	Total "W" lbs.	Stress (psi) "S"	Cycles to Failure "N"
1	10	43	53	40,000	72,000
2	10	43	53	40,000	88,000
3	10	40	50	37,700	140,000
4	10	36	46	34,700	300,000
5	10	33	43	32,400	500,000
6	10	30	40	30,200	810,000
7	10	26.5	36.5	27,500	4,400,000
8	10	23	33	25,000	8,200,000
9	10	22	32	24,200	30,000,000
10	10	18	28	21,000	82,000,000

Stress = 755 x W

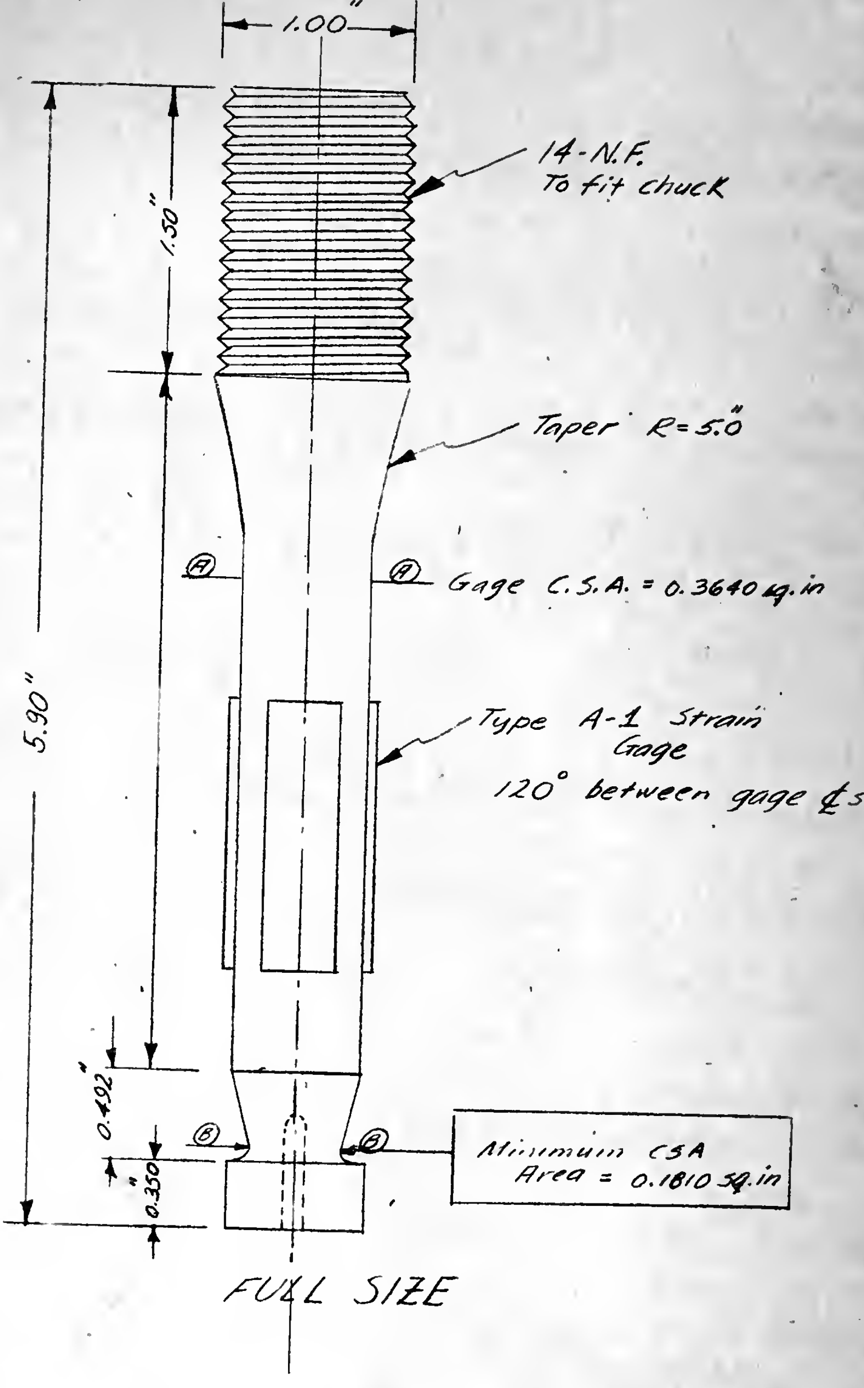
TABLE IV

STRESS CONCENTRATION FACTORS FOR PROPELLER BLADE MODELS

$$\text{SCF} = \frac{\text{Max. stress}}{\text{F/A}}$$

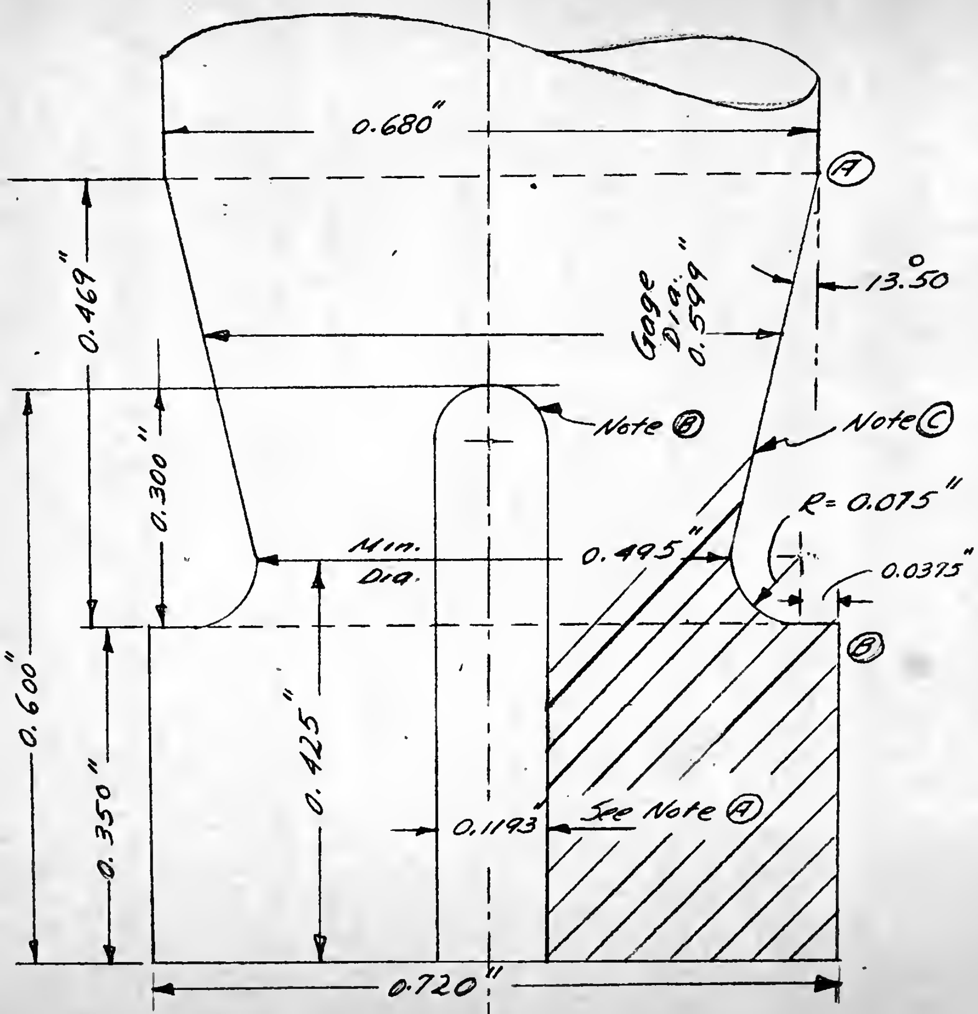
Specimen No.	Cycles to Failure	Nominal Stress (F/A)	Apparent Max. Stress	SCF
1	30,000	27,900	54,000	1.935
2	64,000	27,800	50,500	1.816
3	91,000	27,020	47,750	1.767
5	175,000	25,080	45,000	1.794
4	257,000	22,380	43,500	1.943
8	300,000	21,990	43,000	1.955
9	400,000	21,760	41,600	1.911

Figure 1. Failure of Wind Tunnel Propeller Blade.



FATIGUE SPECIMEN
Showing gage locations

Figure 2



FATIGUE SPECIMEN

Ref: Ham. Std. Dwg. #SK16000
(SCALE: x5)

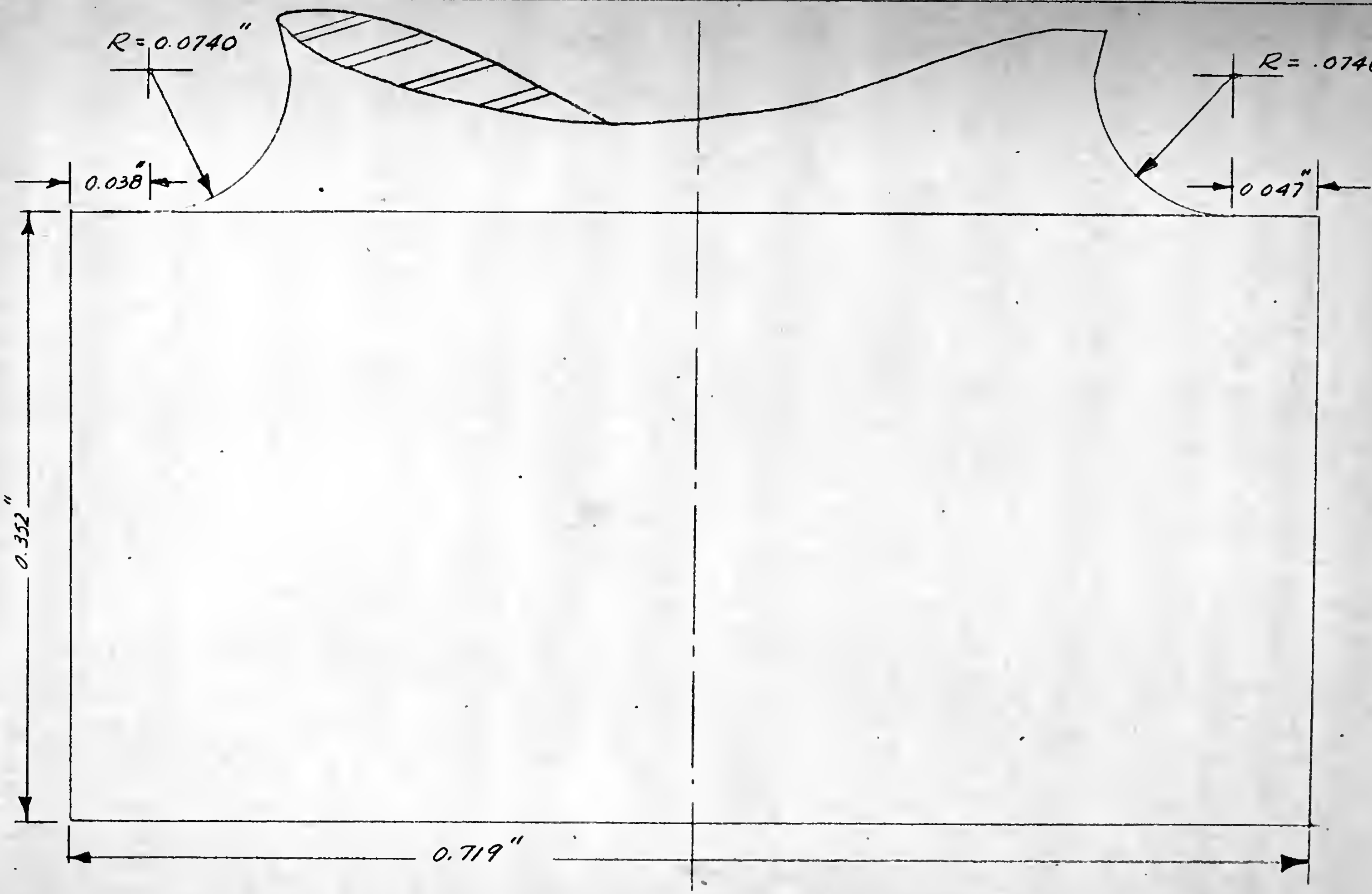
NOTES:

(A) Drill with #32 and hand lap.

(B) Central hole spherical at bottom

(C) Finish surface to 5μ inches. between points A and B

Figure 3



Dimension Check with
Jones & Lamson Optical Comparator
Figure 4

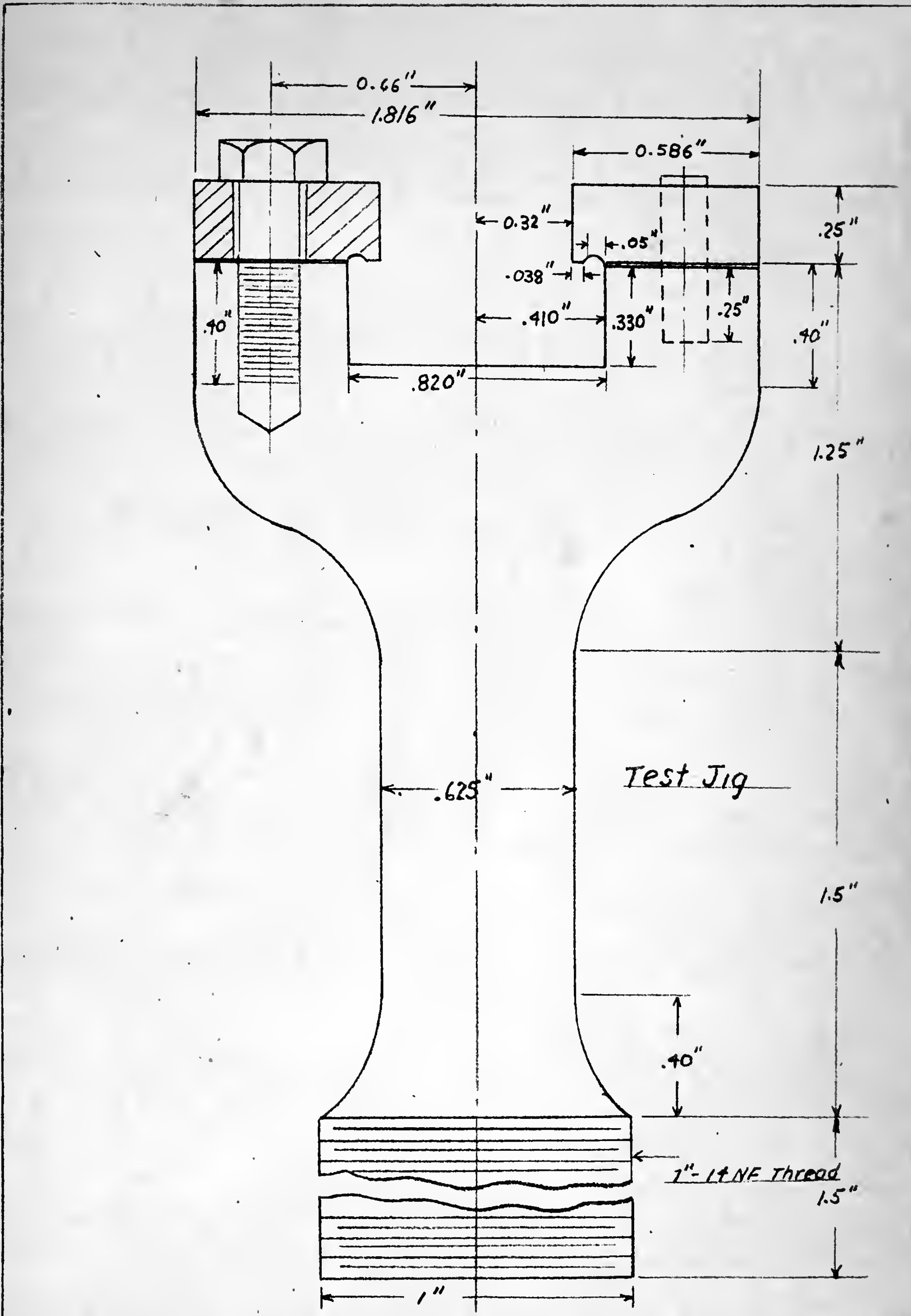

0.66"
1.816"
0.586"
0.32"
.05"
.25"
.038"
.410"
.330"
.25"
.40"
.40"
.820"
1.25"
Test Jig
.625"
1.5"
.40"
1"- 14 NF Thread
1.5"
1"

Fig. 5

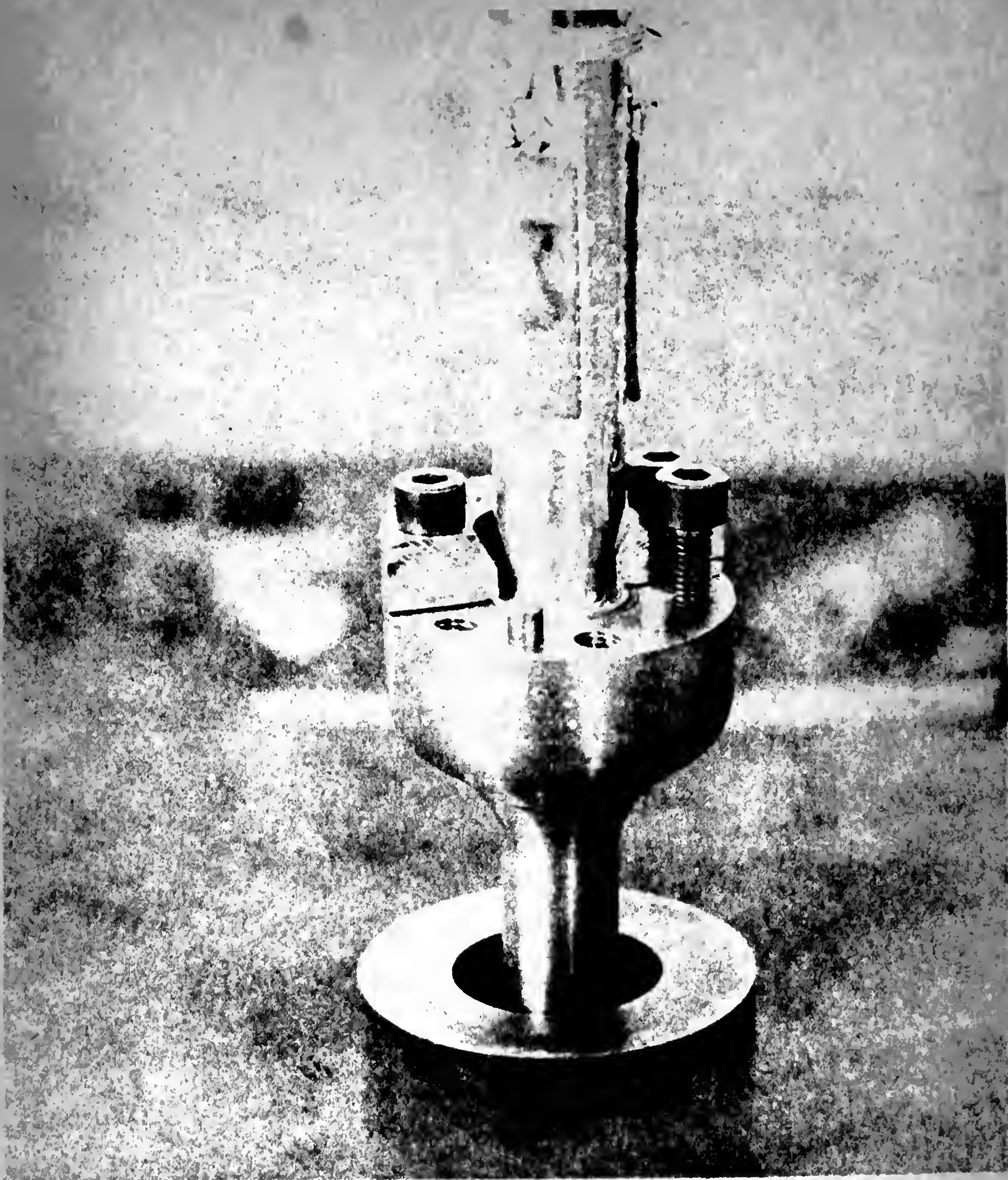


Figure 6. Specimen Mounted in Holding Jig.

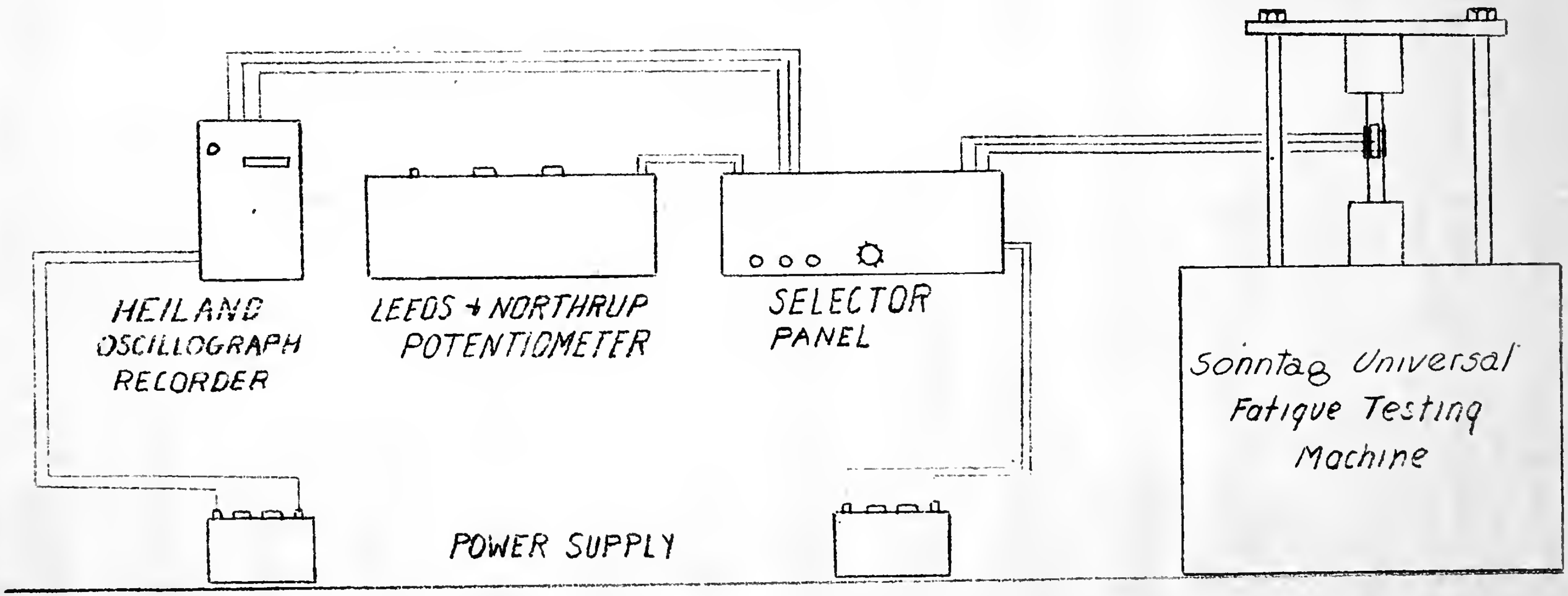


Fig. 7
Schematic Diagram
of
Test Equipment

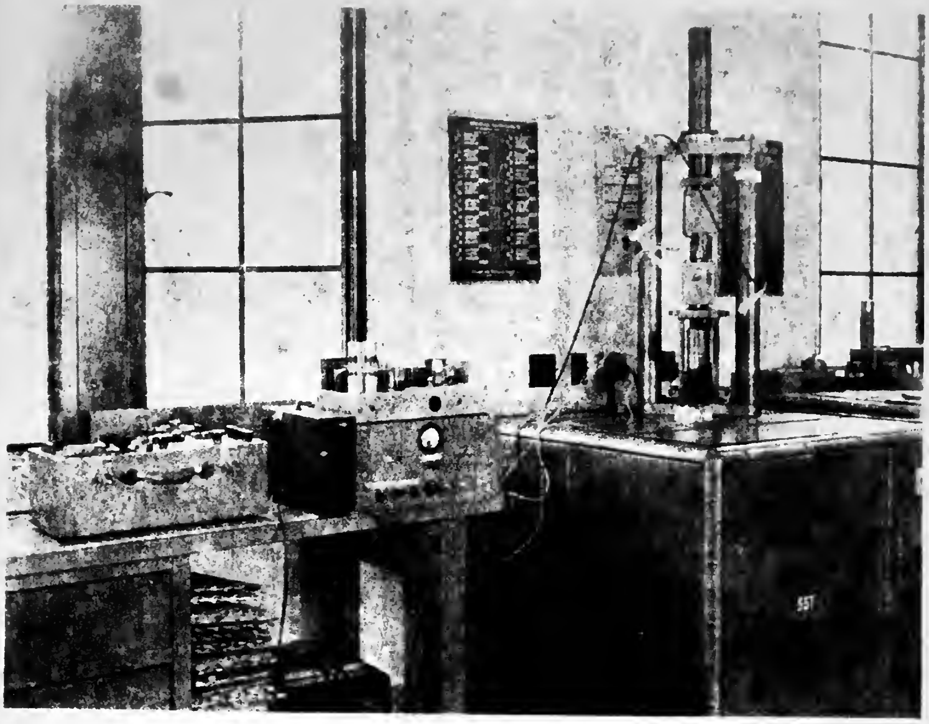


Figure 8. Test Set-up.

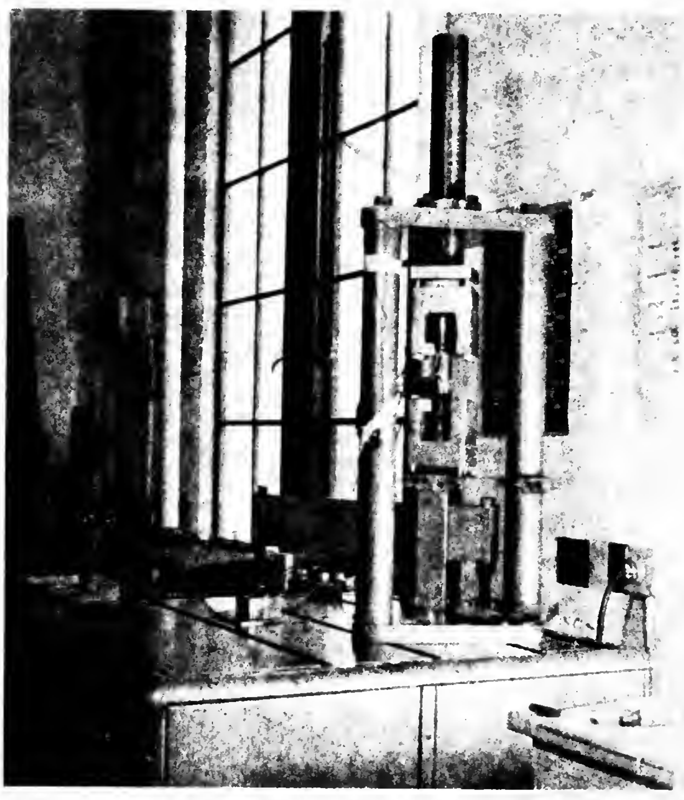


Figure 9. Specimen Mounted in Multiplying Fixture

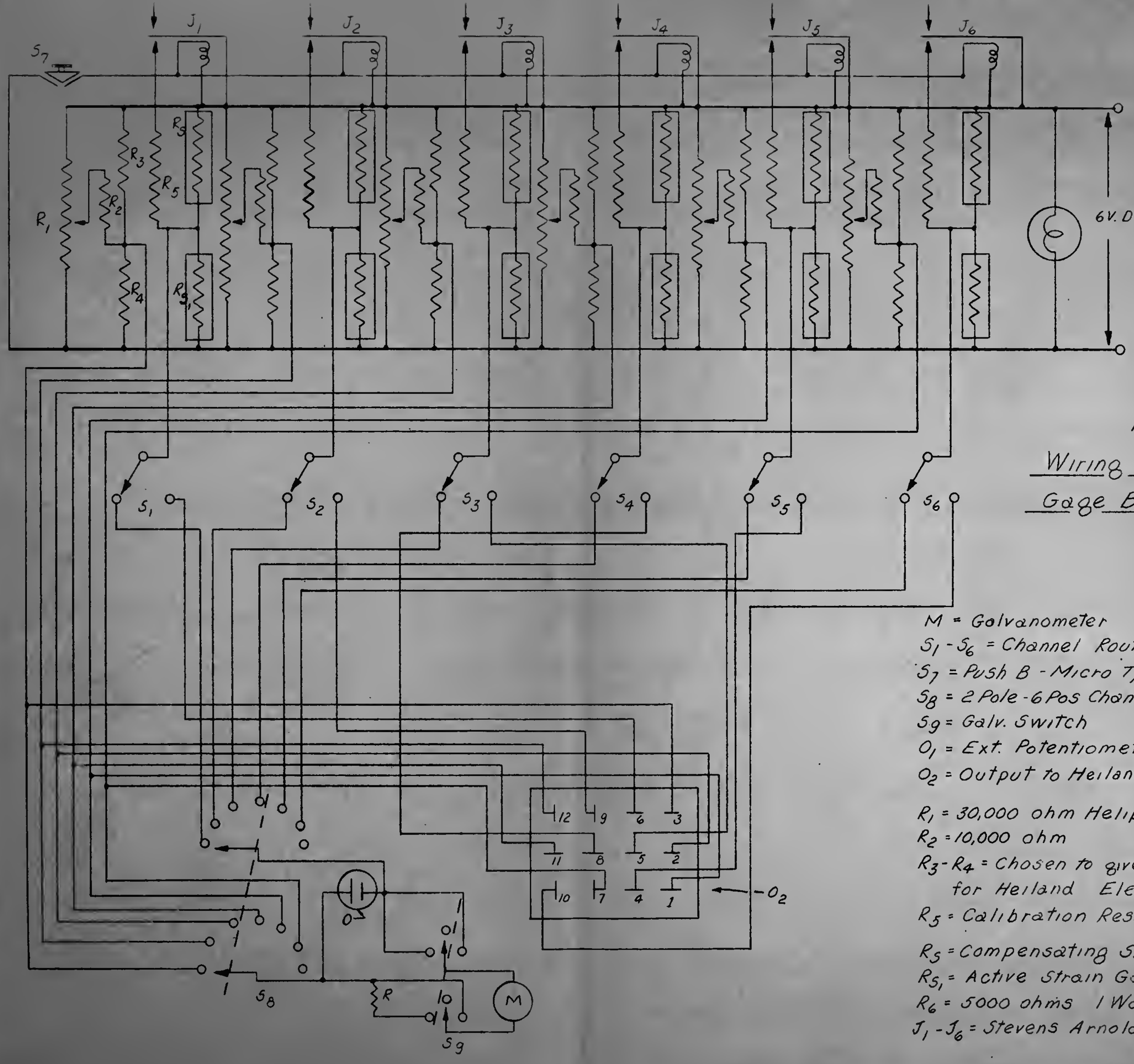


Fig. 10

Wiring Diagram of Strain Gage Balance Panel

M = Galvanometer

S_1 - S_6 = Channel Routing

S_7 = Push B - Micro Type (Norm. Open)

S_8 = 2 Pole - 6 Pos Channel Selector

S_9 = Galv. Switch

O_1 = Ext. Potentiometer

O_2 = Output to Heiland

R_1 = 30,000 ohm Helipot
R_2 = 10,000 ohm
} Same on all channels

R_3 - R_4 = Chosen to give critical damping for Heiland Element used

R_5 = Calibration Resistor

R_S = Compensating Strain Gage

R_{S_1} = Active Strain Gage

R_6 = 5000 ohms 1 Watt

J_1 - J_6 = Stevens Arnold Millisec Relay #172 (Norm. Open)

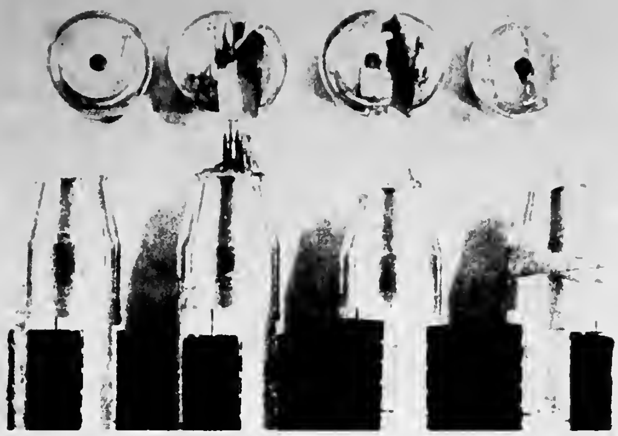


Figure 11. Specimens After Failure. (Top View).

Figure 12. Specimens After Failure. (Side View).

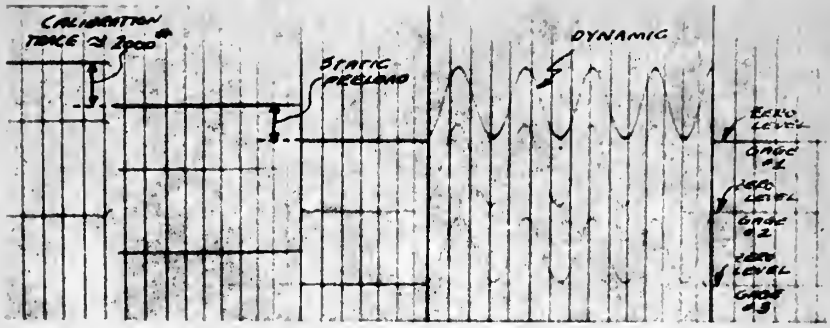


Figure 13 -- Typical Heiland Record

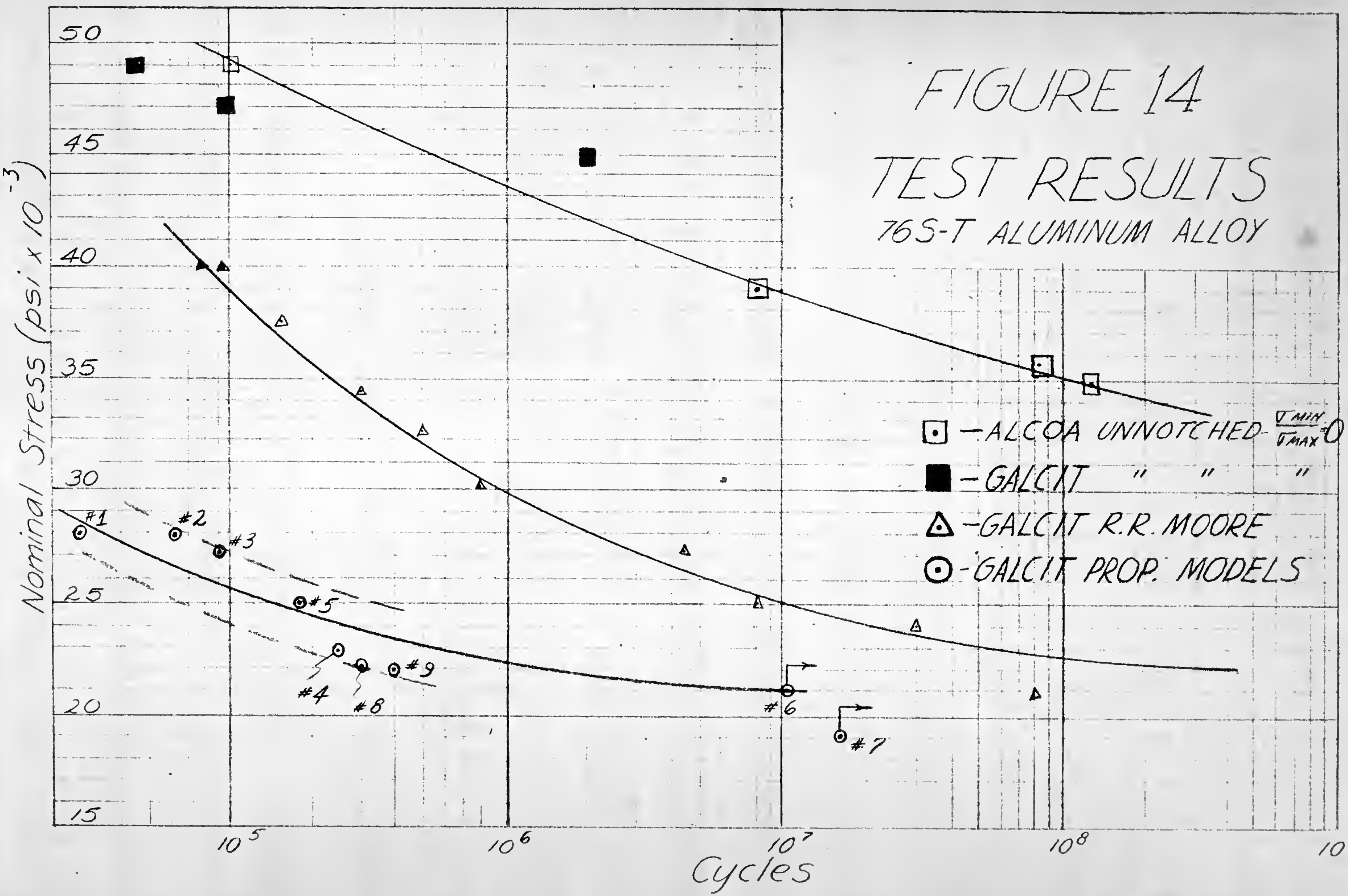


FIGURE 14
TEST RESULTS

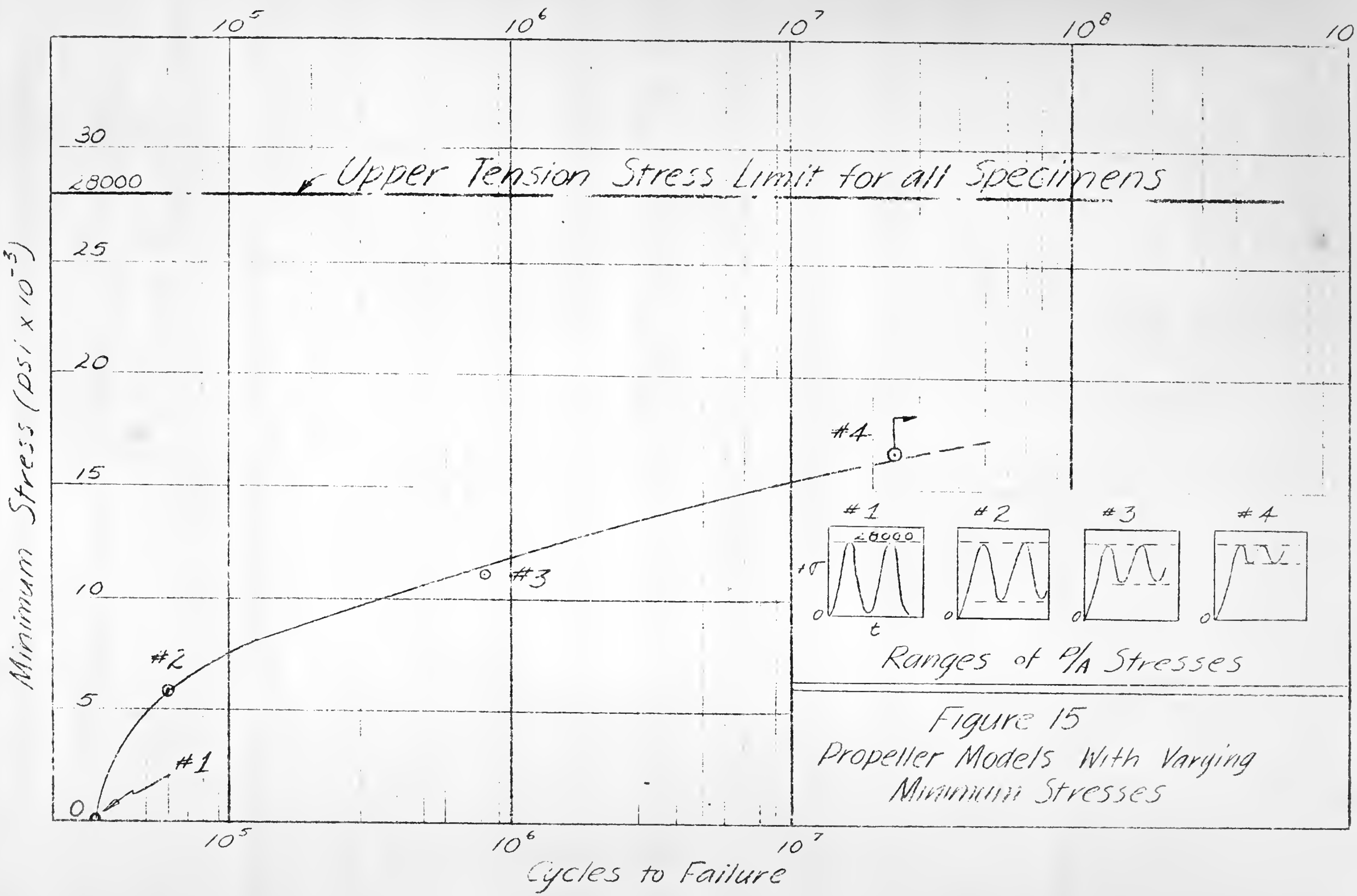
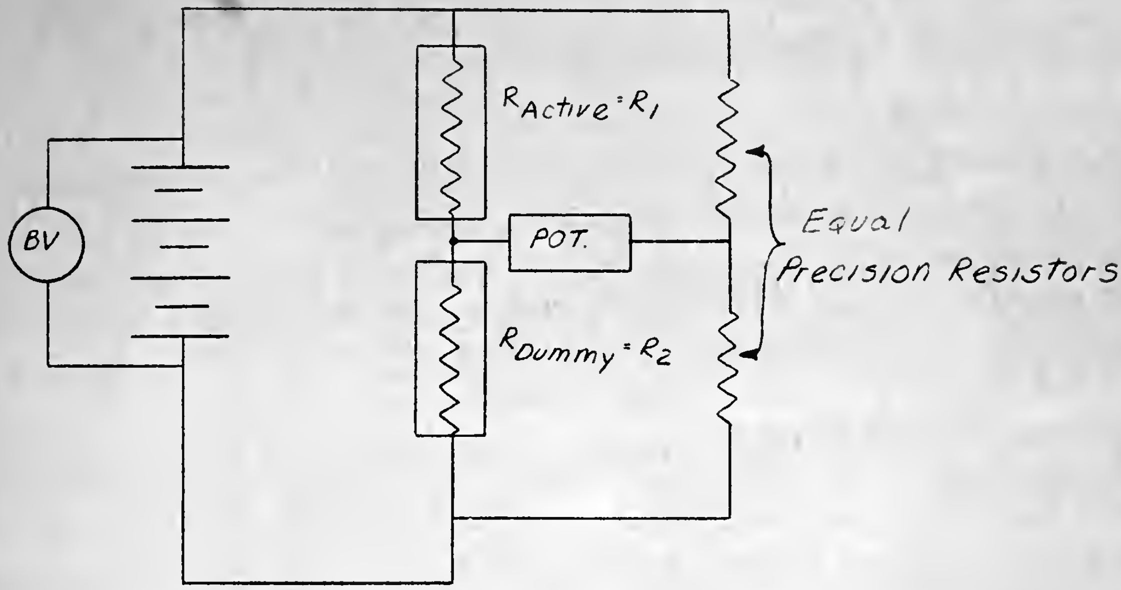


Figure 15
Propeller Models With Varying Minimum Stresses

APPENDIX I - STRAIN FORMULA



Symbols:

(1) R_i = Original gage resistances (no load)

(2) R_i' = Gage resistances after loading

(3) I_i = Original gage currents (no load)

(4) I_i' = Gage currents after loading

(5) E_i = Voltages across gages (no load) $(i = 1,2)$

(6) E_i' = Voltages across gages after loading $(i = 1,2)$

(7) ε = Strain (inches per inch)

(8) $\pm \Delta E$ = Voltage change across active gage after loading as measured by potentiometer

Derivation:

$$E_1 = E_2$$

$$GF = \frac{\frac{\Delta R}{R}}{\frac{\Delta L}{L}} = \frac{\Delta R_1}{R_1 \varepsilon}$$

$$I_1' = \frac{E_1'}{R_1'}$$

$$I_2' = \frac{E_2'}{R_2}$$

$$E_1' = E_1 \pm \Delta E$$

$$E_2' = E_2 \mp \Delta E$$

$$I_2' = I_1'$$

Derivation:

$$\Delta R_1 = R_1' - R_1$$

$$\frac{E_1 \pm \Delta E}{R_1'} = \frac{E_2 \mp \Delta E}{R_2}$$

$$R_2 E_1 \pm R_2 \Delta E = (R_1 + \Delta R_1)(E_2 \mp \Delta E)$$

$$\pm R_2 \Delta E = \Delta R_1 E_2 \mp R_1 \Delta E \quad \text{(Eliminating second order terms)}$$

$$\pm \Delta E (R_1 + R_2) = \Delta R_1 E_2 \qquad \Delta R_1 = (GF)(R_1)(\mathcal{E})$$

$$\pm 2\Delta E = (GF)(\mathcal{E})(E_2)$$

$$\mathcal{E} = \pm \frac{2\Delta E}{(GF)(E_2)}$$

$$\text{but } E_2 = \frac{\text{Battery Voltage}}{2}$$

$$\mathcal{E} = \frac{\pm 4\Delta E \text{ (Volts)}}{\text{(Gage Factor)(Battery Voltage)}} = \frac{\pm (4 \times 10^{-3}) \text{ Millivolts}}{(GF)(BV)}$$

Thesis 12998
S443 Shepherd
Correlation of fatigue data to determine stress concentration factors in 76S-T aluminum alloys.